Eliminating Multiple Modes of Vibration in a Flexible Manipulator

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Abstract

The flexibility of long reach manipulators presents a difficult control problem when accurate end-point position is required. To maintain a desired tip trajectory, the residual vibration inherent to flexible systems must be eliminated. Unfortunately, vibration suppression is often difficult to achieve in systems whose parameters are a function of configuration. This paper discusses a modified command filtering technique that eliminates the first two modes of vibration in a large, flexible manipulator with position dependent parameters. The elimination of residual vibration is demonstrated using a circular trajectory located in the workspace of the manipulator so that a change in system properties occurs. The modified command filtering technique will be compared to previous control methods to demonstrate its effectiveness in eliminating residual vibration.

Introduction

A recent development in the control of flexible manipulators is preshaping commanded inputs to reduce the residual vibration of the system. Work conducted by Meckl and Seering demonstrated that force profiles can be developed to control flexible dynamic systems with minimal vibration. By defining an appropriate cost function, a force profile can be derived that efficiently allocates kinetic energy so that excitation is minimized at the system resonances and maximum energy is used for system motion [1]. Recently, Meckl and Kudo applied the minimum energy force profile technique to control the position of disk drive heads [2]. Using a finite series of ramped sinusoids, simulation results showed significant reduction in vibration at the head positioner natural frequency. However, they concluded that the feedforward nature of the control would likely produce final positioning errors that would require the use of an existing closed-loop feedback system.

Another feedforward preshaping technique is the impulse shaping method developed by Singer and Seering. Their method transforms each sample of the desired input into a new set of impulses that do not excite the system resonances [3]. The idea involves delaying a portion of the input by half the damped natural period of the system to cancel the vibration induced by the original input. The method was implemented strictly in a feedforward manner and depended heavily on accurate system information to be effective.

Some interesting extensions of the impulse shaping method have recently been considered. Hillsley and Yurkovich designed a two stage control architecture to achieve accurate end-point position control for point-to-point movements [4]. The first stage implements Singer and Seering's shaping algorithm to achieve vibrationless motion for large angle slewing and then shaft position, end-point acceleration feedback accurately positions the tip. This composite control scheme provides vibration suppression during gross motions as well as reducing overshoot at the final end-point position. Rattan and Feliu presented the design of a feedforward controller using a dynamic model inversion technique [5]. Using a simplified discrete-time controller that is independent of the reference input, a continuous controller is derived that contains a finite sum of delay terms. In fact, if the delay period is chosen to equal one-half the damped natural period, a shaping controller similar to Singer and Seering's impulse shaping method results.

The shaping method can also be applied directly in a feedback control architecture. Noakes, Petterson and Werner applied an acceleration profile to damp oscillations in objects transported by overhead cranes [6]. By delaying a constant acceleration input by one-half the period of the swinging object, the object is able to move with constant velocity through the workspace. The same strategy is applied to decelerate the object and bring it to a stop without overshoot. Magee and Book have also used a modified command filtering scheme in a feedback manner to eliminate the first mode of vibration in a two-link, flexible manipulator named RALF (Robotic Arm, Large and Flexible). They demonstrated that the original impulse shaping scheme developed by Singer and Seering induces vibration in time-varying systems and then created a modified command filtering technique to eliminate the undesirable motion [7,8]. The modified command filtering technique alters the feedback error in a P.D. control scheme to achieve significant vibration suppression in a time-varying system.

The main focus of this paper is the application of modified command filtering to eliminate the first and second modes of vibration of RALF. A filter for each mode of vibration is designed and then combined to eliminate two modes simultaneously. Experimental results verify the elimination of residual vibration using the modified command filtering technique when compared to impulse shaping and an original P.D. feedback control scheme.

Simplified model of system

A previous modelling technique to describe the dynamic behavior of RALF utilizes an assumed modes method to approximate the position of the manipulator and then applies Lagrange's equation of motion. This procedure yields a set of nonlinear equations which couples the rigid body and flexible motion of the system. A more elementary approach can be taken to model the two modes of vibration. The flexible motion is assumed to be a linear combination of two second-order systems with the overall transfer function

$$H(s) = \frac{\omega_{n1}^2 \cdot \omega_{n2}^2}{(s^2 + 2\zeta_1 \omega_{n1} s + \omega_{n1}^2)(s^2 + 2\zeta_2 \omega_{n2} s + \omega_{n2}^2)} \qquad (1)$$

which relates position to input force.

Since the two natural frequencies and damping ratios are time-varying, these parameters were determined experimentally to achieve accurate results. The manipulator was stimulated with random noise at specified joint configurations throughout the workspace and then the frequency response of the system was taken for each configuration using a digital Fourier analyzer. From the frequency response data, the natural frequency and damping ratio can be calculated and then parameterized as a function of joint configuration. Thus, the natural frequency and damping ratio for any joint configuration can be computed when needed by the modified command filtering method.

Modified command filtering

The modified command filtering technique is a time-varying filter that accommodates changes in the system parameters. It contains filter coefficients that are similar to the impulse shaping method and makes modifications to the impulse output when a change in discrete-time period occurs. Consider a filter that creates three output impulses for each sample of the input. The continuous-time filter can be written in the form

$$G_1(s) = A_{10} + A_{11} e^{-s\,delT_1} + A_{12} e^{-s\,2\,delT_1} \qquad (2)$$

where the continuous-time period, $delT_1$, is one-half the damped natural period of the system and the coefficients, A_{ij}, can be found in [9]. The continuous-time period can be transformed into a discrete-time period, $deln_1$, using the sampling rate of the control system. The equation to perform this transformation is

$$deln_1 = int(delT_1 \cdot f_s) \qquad (3)$$

where f_s is the sampling rate of the control system and the $int(\cdot)$ function truncates the argument to an integer.

For time-invariant systems, the discrete-time period, $deln_1$, is constant even though the damped natural frequency may not be known precisely. No problems arise for the impulse shaping method for this situation. However, if the damped natural frequency becomes time-varying, the continuous-time period, $delT_1$, becomes time-varying as well. A significant change in continuous-time period will generate a change in discrete-time period which produces a vibration in the system. This induced vibration is verified later in the paper.

The modified command filtering method accommodates a change in discrete-time period and prevents vibration from being induced into the system. For each discrete-time sample of the desired command, the modified method compares the current discrete-time period to the previous discrete-time period. If the discrete period has increased, it is as if the discrete-time scale has expanded leaving gaps in the filter output. To fill the gaps, the current sample is filtered twice using both discrete-time periods and adding the extra impulses at the appropriate discrete-time locations. If the discrete-time period decreases, it is as if the discrete-time scale is compressed leaving extra impulses in the filter output. These extra impulses are subtracted from the filter output to accommodate the compressed discrete-time scale. It is worth noting that the extra impulses subtracted for a decrease in discrete-time period correspond to the same impulses that were added for the increase in discrete-time period [7,8,9].

Two mode filter

The continuous-time filter given in Equation (2) can be implemented for a second mode of vibration and used to create a filter that eliminates two modes simultaneously. The second filter takes the form

$$G_2(s) = A_{20} + A_{21} e^{-s\,delT_2} + A_{22} e^{-s\,2\,delT_2} \qquad (4)$$

where the subscript '2' signifies the second mode. To form the two mode filter, the independent filters are multiplied together which results in convolution of the coefficients. The resulting filter can be written as

$$G(s) = G_1(s) \cdot G_2(s) \tag{5}$$

and then expanded to yield

$$\begin{aligned} G(s) = \; & A_{10} \cdot A_{20} + A_{10} \cdot A_{21} e^{-s\,delT_2} + A_{11} \cdot A_{20} e^{-s\,delT_1} \\ & + A_{10} \cdot A_{22} e^{-s\,2\,delT_2} + A_{11} \cdot A_{21} e^{-s(delT_1 + delT_2)} \\ & + A_{12} \cdot A_{20} e^{-s\,2\,delT_1} + A_{11} \cdot A_{22} e^{-s(delT_1 + 2\,delT_2)} \\ & + A_{12} \cdot A_{21} e^{-s(2\,delT_1 + delT_2)} + A_{12} \cdot A_{22} e^{-s(2\,delT_1 + 2\,delT_2)}. \end{aligned} \tag{6}$$

The critical part in the design of a multiple mode filter is to now renormalize the exponential coefficients so that they still sum to one. This normalization ensures that the overall amplitude of the new impulse sequence is the same as the amplitude of the desired input sample. It is the same normalization used in the design of each filter individually. The renormalized filter can be written as

$$\begin{aligned} G(s) = \; & G_0 + G_1 e^{-s\,delT_2} + G_2 e^{-s\,delT_1} \\ & + G_3 e^{-s\,2\,delT_2} + G_4 e^{-s(delT_1 + delT_2)} \\ & + G_5 e^{-s\,2\,delT_1} + G_6 e^{-s(delT_1 + 2\,delT_2)} \\ & + G_7 e^{-s(2\,delT_1 + delT_2)} + G_8 e^{-s(2\,delT_1 + 2\,delT_2)} \end{aligned} \tag{7}$$

where the new coefficients become

$$G_0 = \frac{A_{10} \cdot A_{20}}{DEN} \tag{8}$$

$$G_1 = \frac{A_{10} \cdot A_{21}}{DEN} \tag{9}$$

$$G_2 = \frac{A_{11} \cdot A_{20}}{DEN} \tag{10}$$

$$G_3 = \frac{A_{10} \cdot A_{22}}{DEN} \tag{11}$$

$$G_4 = \frac{A_{11} \cdot A_{21}}{DEN} \tag{12}$$

$$G_5 = \frac{A_{12} \cdot A_{20}}{DEN} \tag{13}$$

$$G_6 = \frac{A_{11} \cdot A_{22}}{DEN} \tag{14}$$

$$G_7 = \frac{A_{12} \cdot A_{21}}{DEN} \tag{15}$$

$$G_8 = \frac{A_{12} \cdot A_{22}}{DEN} \tag{16}$$

and the renormalizing denominator is

$$\begin{aligned} DEN = \; & A_{10} \cdot A_{20} + A_{10} \cdot A_{21} + A_{11} \cdot A_{20} \\ & + A_{10} \cdot A_{22} + A_{11} \cdot A_{21} + A_{12} \cdot A_{20} \\ & + A_{11} \cdot A_{22} + A_{12} \cdot A_{21} + A_{12} \cdot A_{22}. \end{aligned} \tag{17}$$

Now that the two mode filter has been designed, it can be implemented in a P.D. feedback control structure to eliminate unwanted residual vibration.

Feedback control structure

To prevent end-point positioning errors, the modified command filtering method and the impulse shaping method are applied to each sample of the error term in a P.D. feedback control system. Figure 1 shows the block diagram of the control structure used to suppress residual vibration in RALF.

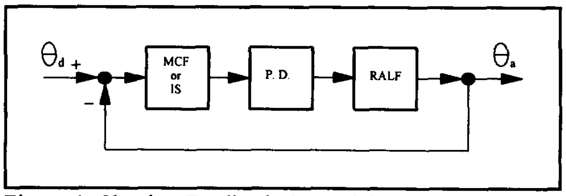

Figure 1. Shaping-Feedback Control System

This particular implementation differs from some other control strategies when using a feedback control scheme with a shaping method. Singer and others place the shaping filter before the feedback loop and only filter the desired response. One advantage of this new implementation is that the experimental data, i.e. natural frequency and damping ratio, can be used directly in the filtering algorithm and the effective natural frequency and damping ratio of the feedback control system do not have to be calculated.

Desired trajectory

To compare the modified command filtering method to other control methods, a test trajectory must be generated that contains specific frequency components to allow a fair comparison. The desired path is a 3-ft. diameter circle located in the manipulator's workspace where a deviation in system parameters occurs. The deviation is less than 10% in natural frequency and is less than 100% in damping ratio for either mode. This large deviation in damping ratio is representative of the inability to accurately measure this quantity. To artificially excite the manipulator, two sinusoidal components are added to the radial component of the circle with frequencies corresponding to the first two modes of vibration of the

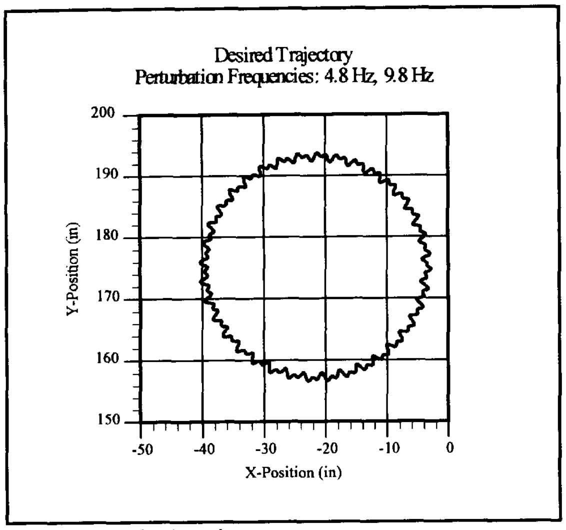

Figure 2. Desired Trajectory

system. The test trajectory is shown in Figure 2 and the added sinusoidal components are very apparent. This trajectory is a worst case scenario and will demonstrate the effectiveness of the vibration suppression methods.

Experimental results

In the two mode filter design example, each sample of the input generated three impulses for each mode. To increase robustness to variations in system parameters, four output impulses are used for each mode in these experiments. The advantage of increasing the number of impulses can be seen in Figure 3.

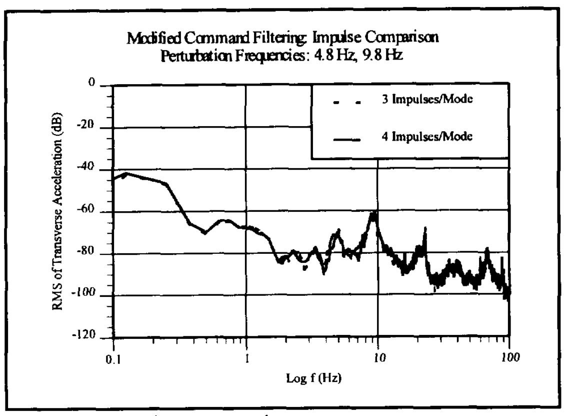

Figure 3. Impulse Comparison

The frequency response of an accelerometer mounted at the tip of the manipulator compares the modified command filtering method when three and four impulses are used for each mode. The difference in the frequency responses is nearly 3 dB at the first mode of vibration and 1 dB at the second mode of vibration.

The second experiment compares the acceleration frequency response of a one mode filter compared to a two mode filter when the desired trajectory is input into the shaping-feedback control system. Figure 4 shows the effectiveness of the two mode filter. At the second natural frequency (10 Hz), the difference in frequency response is 31.5 dB. The acceleration amplitude of vibration for the two mode filter is 0.07% of the amplitude for the single mode filter. Another point worth noting is the decrease in performance at the first mode of vibration for the two mode filter. The difference is only 2.7 dB but the filter is less effective at the first mode of vibration when multiple mode suppression is desired.

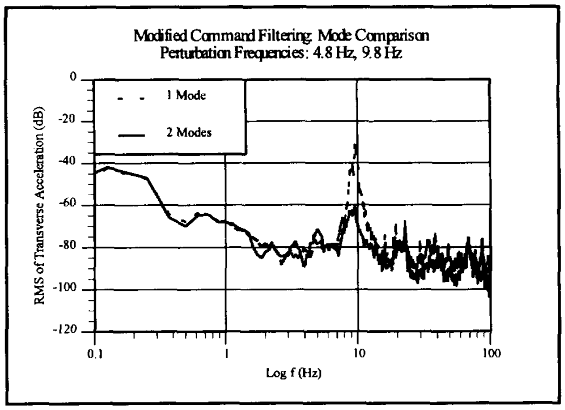

Figure 4. Mode Comparison

The third comparison is between modified command filtering and impulse shaping to eliminate two modes of vibration when the desired trajectory is fed into the shaping-feedback control structure. The system parameters vary with time for each method which causes the impulse shaping method to induce a vibration into the system. Figure 5 shows the frequency response comparison for this experiment. Notice that the impulse shaping method induces a 20 Hz vibration into the system that is 33 dB larger than the modified command filtering method. This difference results in a vibration that is 2000 times larger in acceleration amplitude than what is produced by the modified filtering method.

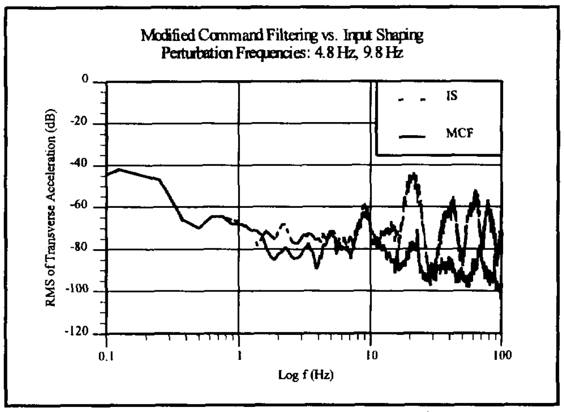

Figure 5. Time-Varying Parameter Comparison

The impulse shaping method can produce more favorable results if the time-varying system parameters are averaged and constant values are used. Figure 6 shows the frequency response comparison when constant parameters are used. Notice that the modified command

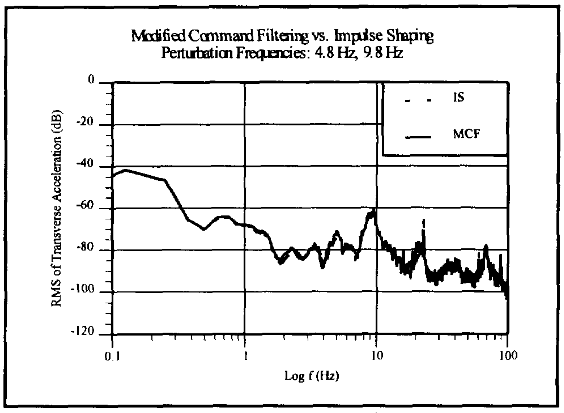

Figure 6. Constant Parameter Comparison

filtering method using time-varying parameters and the impulse shaping method using constant parameters produce comparable results. The reason for the equivalent performance is because the deviations in natural frequency and damping ratio are within tolerable ranges for the impulse shaping method. Trajectories that contain large variations in system parameters are currently being investigated.

The last experiment compares the modified command filtering method for two modes of vibration with the original P.D. control routine. Figure 7 displays the frequency response comparison for the two control schemes. It is evident that the P.D. control routine alone cannot accommodate inputs that contain resonant frequencies of the manipulator. The difference in frequency response at the first mode is 32.4 dB and nearly 34 dB at the second mode. The P.D. feedback control scheme is effective at eliminating final positioning errors but does not prevent the residual vibration in a flexible manipulator.

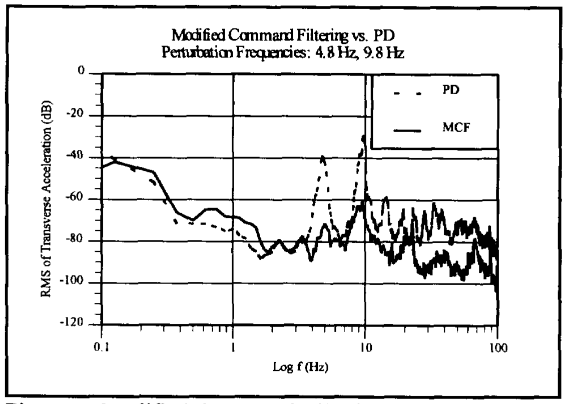

Figure 7. Modified Command Filtering vs. PD

Conclusions

The elimination of the first two modes of vibration in a time-varying system was presented using both the modified command filtering method and the impulse shaping method. The two mode filter needed to implement the methods was derived and the renormalization of the filter coefficients was discussed. The robustness of using extra impulses for each mode was considered and the effectiveness of adding the second mode filter examined. The impulse shaping method was shown to induce vibration if the system parameters are allowed to vary with time but produced comparable results with the modified command filtering method if the parameters are held constant. It is believed that the modified command filtering method will outperform the impulse shaping method for large variations in system parameters and is currently being investigated.

Acknowledgements

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